Exergy evaluation and ORC use as an alternative for efficiency improvement in a CI-engine power plant

da Silva, Julio; Seifert, V; de Morais, V; Tsolakis, Athanasios; Herreros, Jose; Torres, Emilio

DOI:
10.1016/j.seta.2018.10.007

Exergy Evaluation and ORC Use as an Alternative for Efficiency Improvement in a CI-Engine Power Plant

Julio A. M. da Silva, julio.silva@ufba.br
Department of Mechanical Engineering, Federal University of Bahia, Salvador, Bahia, Brazil
Vitor Seifert, vrseifert@hotmail.com
Industrial Engineering Program, Federal University of Bahia, Salvador, Bahia, Brazil
Vinícius O. B. de Morais , viniciusmorais_91@hotmail.com
Department of Mechanical Engineering, Federal University of Bahia, Salvador, Bahia, Brazil
Athanasios Tsolakis, a.tsolakis@bham.ac.uk
School of Mechanical Engineering, University of Birmingham, Edgbaston, UK
Jose Herreros, j.herreros@bham.ac.uk
School of Mechanical Engineering, University of Birmingham, Edgbaston, UK
Ednildo Torres, ednildotorres@gmail.com
Department of Chemical Engineering, Federal University of Bahia, Salvador, Bahia, Brazil

Abstract. *The search for efficient processes in order to reduce the dependence on fossil fuels and to mitigate pollutants emission is mandatory in current society. Exergy analysis provides a clear indication of where efforts should be concentrated in the search for more efficient processes. Once the processes responsible for main exergy destruction are revealed, new technologies can be used to decrease the entropy generation so that the efficiency of the process is increased. In this paper a Brazilian power plant with installed capacity of 300MW using turbo-aspirated compression ignition (CI) engines is investigated. Results reveal that although about 29% of energy input is sent to environment from low and high temperature cooling systems, it represents only 2% in exergy terms. Furthermore, results show that the highest exergy destruction, 44%, happens inside the engine and it is unavoidable considering the used technology. The exergy of exhaust gases represents about 6.5% of the exergy input and can be recovered for power generation. The use of an organic Rankine cycle (ORC) optimized in respect to working fluid, operation conditions and configuration can recover 45% of the exhaust gases exergy increasing the overall fuel to electricity energy efficiency of the plant from 43.1% to 46.2%.*

Keywords*: ORC, exergy analysis, power plant, CI-Engines, efficiency increase.*

1. INTRODUCTION

The search for renewable energy sources and high energy efficiency in power generation has been a common concern in recent years. According to the new policies scenario of IEA, the global energy needs will expand in 30% by 2040 [1]. In Brazil, hydroelectric, wind and solar power plants represent together more than 71% of installed capacity for electricity production, however the installed capacity of thermal power plants is increasing: 4.6% of increment in 2015 [2] and 4.3% of increment in 2016 [3]. Since many of these thermal power plants had been conceived as a backup for the hydroelectric system, sporadic use (low capacity factor) has been expected. Thus, many of these plants are based on reciprocating engine in simple (opened) cycles. As the use of these plants increases, problems with maintenance arise and the search for higher efficiency increases.

The Exergy Analysis is a method that systematically combines The First and The Second Laws of Thermodynamics in order to calculate the capacity for work generation of a given substance [4–8]. It is regarded as a powerful tool to locate the main sources of irreversibility in processes involving thermal and chemical energy conversion. Hence, attention can be paid on the most irreversible processes and new technologies can be tested to overcome the problem.

Some works are found in the literature regarding the use of reciprocating engines exhaust gases and jacket water to generate power. The exhaust gases can also be used to supply the energy required in catalytic fuel reforming reactions, endothermic in nature, for the production of hydrogen rich gas, known as reformate. The reformate is then used in the combustion chamber of the engine to increase its efficiency and to reduce pollutants emission such as NOx and particulate [9–14]. However, the temperature required for these reactions is relatively high, about 700ºC, and the presence of oxygen in the exhaust gases of the diesel engine will limit the efficiency of the process and hence the potential for energy recovery. Another way to make use of the exhaust gases to increase overall efficiency of an engine is by coupling a Kalina cycle to it. Different configurations for a Kalina cycle coupled to a large marine engine were tested in [15] and the efficiencies of the proposed Kalina cycles ranged from 20.8% to 23.2%. Yet, it was stated in [16] that organic Rankine cycles (ORC) perform better than Kalina cycles in temperatures higher than 190ºC. The use of a Kalina cycles instead of an ORCs for diesel engine exhaust gases energy recovery is also disregarded in [17] due to the high complexity of the Kalina cycles and the insignificant gain in efficiency in respect to ORCs. Organic Rankine cycle seems to be a promising technology to reduce fuel consumption in heavy-duty diesel engines. The use of an ORC to recover the energy from the exhaust gases of an engine provides a power increase that is dependent on the engine operating conditions. In [18] it was reported that the power increase was 2.12% and 2.74% at 1500 and 2100 RPM, respectively [18]. A review focusing on the use of ORC in vehicles in order to meet the increasingly stringent legislation is found in [19], in which a large potential for reduction in fuel consumption is identified and the ORC technology is identified as one of the most promising technologies allowing fuel economy of up to 10%. The use of ORC to recover the engine exhaust gases energy at full and partial loads is analyzed in [20], improvements of about 6% in efficiency of the overall system are revealed. A power increment of 5.6% was verified by [21] when and ORC was used to recover diesel engine exhaust gases energy. A heavy duty diesel engine coupled with an ORC was evaluated using The First Law of Thermodynamics in [22] and it was stated that a reduction of 10% in fuel consumption could be achieved. An exergy-based optimization was carried out to optimize the exergy efficiency and power output of an ORC coupled to an internal combustion engine in [23], it was verified that the system is very sensible to the efficiency of the dual expander used and the ORC exergy efficiency was found to be around 21%. Yet, efficiencies of only 2 to 3% for a ORC coupled to a reciprocating internal combustion engine was found in [24]. An ORC generating up to 125 kW using the energy of the jacket water of a ship engine was tested in [25]. A new working fluid with low environmental impact was tested in [26] for an ORC using an engine exhaust gases and jacket water as sources of energy. Two separated ORC systems with R245fa and benzene as the working fluids were designed to utilize the waste heat from both the jacket water and the engine exhaust gases in [27], an efficiency increment of 10.2% for the marine diesel engine was verified. An ORC was optimized to recover the energy of an engine operating on natural in [28], efficiencies ranging from 8.9% to 10.19% for the ORC were found. Cascade-ORCs also have been used to recover energy losses from engines efficiently [29], [30], [31]. An ORC using the exhaust gases of diesel engine was evaluated in [32], the increase observed in the overall efficiency was 0.66%. The use of an ORC in a heavy duty truck engine was proposed in [33], an extra power output of 3.07 kW was verified at full cargo load at 95 km/h and an extra consumption of 0.67 kW was found at 30km/h due to the increase of weight. An extensive review on the use of ORCs coupled to CI engines is found in [34], it was pointed out that the evaporator of the ORC should be design considering the variable characteristics of the exhaust gases. In [18] it was verified that the backpressure increment of the engine with ORC increases with the increasing of ORC cooling water inlet temperature. These problems don't appear to be significant for power plants in which the load of the engine is kept constant and the back pressure can be decreased by a proper chimney project or by the use of blowers/exhausters.

In this work a CI-turbo-aspirated-engine-based power plant is assessed using exergy analysis to highlight the components responsible for the main exergy destruction and losses so that chances for efficiency improvement can be found and quantified. The analysis revealed that the energy lost by the cooling circuits is insignificant for power generation and the exhaust gases leaving the recovery boiler have significant quantity of exergy. Hence, an ORC was suggested and optimized in its working fluid, operating conditions (sub and supercritical) and configuration (with and without recuparator) in order to quantify the benefit that this new technology can bring for the evaluated power plant.

2. EXERGY ANALYSIS METHODOLOGY

Exergy analysis enables the evaluation of the work generation capacity of energy losses, thus it allows comparison of losses in a common thermodynamic basis taking the quality of the energy into consideration. Furthermore, it reveals the processes in which energy is conserved but the capacity to generate work (exergy) is not.

Whenever the kinetic and potential components of exergy are negligible, the total exergy is assessed by summing up its physical and chemical components, $b_{total} = b_{ph} + b_{ch}$. While chemical exergy indicates the capacity for work production by taking the substance to chemical equilibrium with the environment and is a function of substance composition, physical exergy indicates the capacity for work generation by taking the substance to physical (temperature and pressure) equilibrium with the environment [35]. It was considered an environment (reference condition) in which T_0 = 25ºC and P_0 = 101.325 kPa. The equations used to calculate the exergy of the streams follow the literature such as [4], [7], [36]. Equation (1) indicates how the physical exergy is obtained.

$$b_{ph} = h - h_0 - T_0 \cdot (s - s_0) \tag{1}$$

Since the water circuits of the application are closed, no change in the composition of this fluid occurs and its chemical exergy does not influence the analysis. On the other hand, the air and fuel react and the exhaust gases are produced. Since the composition of exhaust gases are different from the environmental air, there is a potential for work production which was calculated using Eq.(2). In this equation $b_{i\ st}$ is the standard chemical exergy, γ is the activity coefficient, which can be considered equal to one for ideal mixtures, and y_i is the molar fraction of each component. The atmosphere composition considered as reference (dead state) for standard chemical exergy calculation is approximately 78% of nitrogen, 21% of oxygen and 1% of argon in dry molar basis and 70% of relative humidity such as in [5].

$$b_{ch\ mix} = \sum_i^n b_{i\ st} + RT_0 \sum_i^n x_i \ln \gamma_i y_i \tag{2}$$

The chemical exergy of industrial fuels can be easily obtained by using φ which is the ratio of chemical exergy to the low heating value (LHV) of the fuel, such as in Eq.(3). The value of φ is obtained using the fuel composition, Eq.(4), as indicated in [37] for liquid fuels containing sulphur. The accuracy of this expression is estimated to be ± 0.38%.

$$b_{ch\ fuel} = \varphi.LHV \tag{3}$$

$$\varphi = 1.0401 + 0.1728 \frac{x_H}{x_C} + 0.0432 \frac{x_O}{x_C} + 0.2169 \frac{x_S}{x_C} \left(1 - 2.0628 \frac{x_H}{x_C} \right) \tag{4}$$

The exergy destroyed is given by Gouy-Stodola equation, Eq.(5), in which $T_0 \cdot S_{generated} = \dot{B}_{destroyed}$ and $\dot{B}_{total} = \dot{m} \cdot b_{total} = \dot{m}\left(b_{ph} + b_{ch}\right)$.

$$\dot{B}_{total\ in} = \dot{B}_{total\ out} + T_0 \cdot S_{generated} \tag{5}$$

3. CASE STUDY

The data presented in this section comes from plant project guide and manuals together with plant sensors and mass and energy balances. The power plant is composed of 120 CI-engines of 2480 kWe each. The engines are turbo-aspired with 2 separated cooling systems and 9 cylinders each. The engines currently consume heavy fuel oil (HFO) as primary fuel. -The low temperature (LT) cooling system is responsible for the cooling of the charge air and lubricating oil while the high temperature (HT) cooling system is responsible for removing the heat from cylinders head and jacket. The engines are grouped in 30 islands containing 4 engines each. 16 of these islands have a recovery boiler with capacity to produce 1500 kg/h of saturated steam at 8 bar. The steam is used to keep the fuel heated (130ºC) during its treatment, storage and prior injection in order to meet injector viscosity requirement (12-18cSt). The compressor increases the pressure of charge air from 1 to 3.8 bar and a cooler reduces the air temperature from 190 to 40ºC. The turbine used to drive the compressor makes use of engine flue gases at 412ºC and 3 bar. Then, these flue gases are directed to the recovery boiler in which their temperature is decreased from 269 to 231ºC. The thermal scheme of an island containing a recovery boiler is shown in Fig.1.

For the sake of simplicity only 1 engine was represented, although streams 6, 7 and 8 represent the flue gases from the other 3 engines.

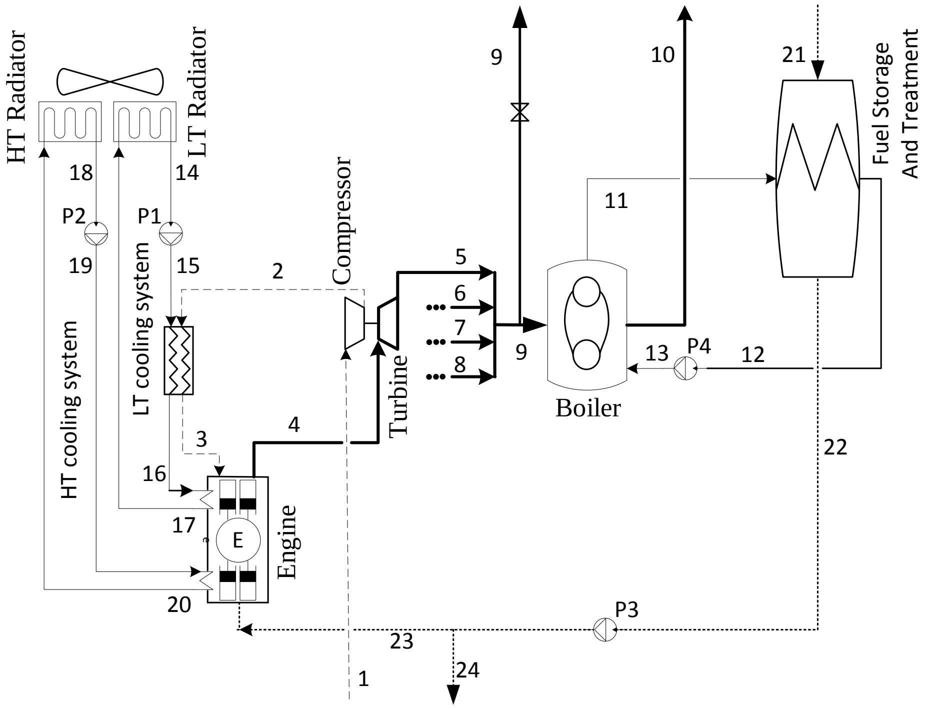

Figure 1. Thermal scheme of an island with recovery boiler

It is worth mentioning that since 16 recovery boilers are used to heat fuel for 120 engines, each recovery boiler is responsible for heating fuel for 7.5 engines. Therefore, the streams 21 and 22 represent the fuel for 7.5 engines while stream 24 represents the fuel for 6.5 engines. The thermodynamic data in Tab. 1 represents the operation condition for thermal schemed presented.

Table 1. Thermodynamic properties of thermal scheme streams

Stream	Fluid	m [kg/s]	T [C]	P [bar]	h [kJ/kg]	s [kJ/kgK]	b_{ph} [kJ/kg]	b_{ch} [kJ/kg]
1	Air	5.22	25.00	1.00	0.00	0.0000	0.00	0.00
2	Air	5.22	190.40	3.80	167.63	0.0625	148.98	0.00
3	Air	5.22	40.00	3.80	15.07	-0.3354	115.07	0.00
4	Flue gas*	5.35	412.12	3.04	433.20	0.6113	250.95	21.41
5	Flue gas*	5.35	269.12	1.03	269.94	0.6529	75.27	21.41
6	Flue gas*	5.35	269.12	1.03	269.94	0.6529	75.27	21.41
7	Flue gas*	5.35	269.12	1.03	269.94	0.6529	75.27	21.41
8	Flue gas*	5.35	269.12	1.03	269.94	0.6529	75.27	21.41
9	Flue gas*	21.42	269.12	1.03	269.94	0.6529	75.27	21.41
10	Flue gas*	21.42	231.18	1.00	222.62	0.5675	53.41	21.41
11	Water	0.42	170.41	8.00	2768.30	6.6615	786.74	50.00
12	Water	0.42	80.00	1.10	334.95	1.0752	18.94	50.00
13	Water	0.42	80.07	8.00	335.83	1.0757	19.68	50.00
14	Water	19.44	36.00	2.00	151.00	0.5186	0.93	50.00
15	Water	19.44	36.02	3.00	151.11	0.5189	0.96	50.00
16	Water	19.44	45.50	2.50	190.74	0.6451	2.97	50.00
17	Water	19.44	50.80	2.20	212.86	0.7140	4.53	50.00
18	Water	8.81	70.00	2.00	293.16	0.9549	13.02	50.00
19	Water	8.81	70.01	3.00	293.27	0.9549	13.12	50.00
20	Water	8.81	82.00	2.50	343.50	1.0990	20.41	50.00
21	Fuel*	1.044	60.00	1.00	108	0.3574	5.99	44152
22	Fuel*	1.044	130.00	1.00	234	0.7007	29.64	44152
23	Fuel*	0.139	130.01	6.00	234.01	0.7007	29.65	44152
24	Fuel*	0.905	130.01	6.00	234.02	0.7008	29.65	44152

* The specific enthalpy indicated for these streams does not include the enthalpy of formation because the variation of enthalpy of formation was replaced by the fuel LHV in the energy balance around the engine (where the combustion occurs).

4. EXERGY EVALUATION OF CASE STUDY

In order to highlight the components in which the capacity to generate work is destroyed an exergy analyses was performed and the results were compared to those from energy analysis. Figure 2 depicts the results from the energy and exergy analysis. The electric power output resents 44.1% of the fuel energy input (plant energy efficiency). The electric power output, however, represents only 40.3% of exergy input since the chemical exergy of the fuel is higher than its LHV ($\varphi = b_{ch}/LHV = 1.069$). The flue gas leaving the boiler represents about 21.2% of the energy input while it represents only 6.5% of exergy input. The same happens to LT and HT radiators, in which a significant quantity of input energy is lost to environment (21.4 e 7.9% of energy input, respectively) while the exergy destroyed in both is about 2.2% of total exergy input. The exergy destroyed in pumps and in the turbo-compressor transmission (TC Trans.) are insignificant for the magnitude used in the analysis.

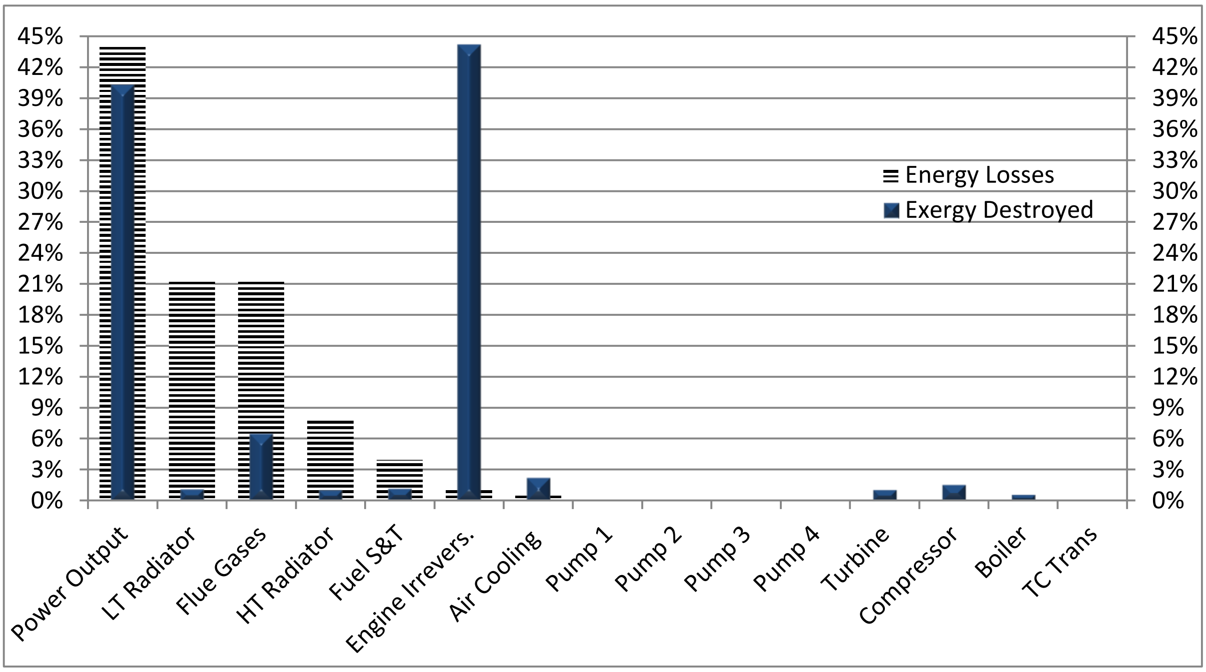

Figure 2. Energy and exergy analysis results for the case study project condition

The exergy analysis highlights that an enormous quantity of exergy (44.2% of input) is destroyed within the engine, even when the losses to cooling systems and exhaust gases are accounted apart. The engine energy loss is very small and comes from heat transfer by convection and radiation to environment, which is calculated using energy balance. On the other hand, the exergy analysis considers all the inefficiencies within this component. The inefficiencies related to combustion process are the main ones and are associated with friction, mixing, occurrence of chemical reaction and heat transfer with finite temperature difference. Most of these inefficiencies are unavoidable for the used technology, but measures can be adopted for minimization [39].

Exergy analysis indicates that the engine cooling water represents only 2.2% of exergy input in this application although this source of energy is sometimes indicated in literature as significant for power generation. Presumably, the addition of further complexity and monetary expenses to recover this energy for power production will provide poor results for this application since the upper limit for power production is the exergy destroyed (sent to environment) in the cooling systems (135 kW) which is attained only for thermal engines with thermal efficiency equals to Carnot efficiency. Exergy analysis also reveals that attention should be paid on the exhaust gases which represent 6.5% of exergy input and a capacity for power generation of 1192kW. Different configurations of optimized ORCs are assessed in the following sections in order to make use of exhaust gases energy to generate power.

5. METHODOLOGY USED FOR OPTIMIZATION OF AN ORC TO RECOVER ENGINE EXHAUST GASES EXERGY

The methodology used for working fluid and configuration selection and for optimization of the ORC operating condition follows [39–41]. It worth mentioning that since the efficiency of the system depends on the proper match between heat source and the working fluid receiving heat as well as its working condition (temperatures, pressures and mass flow), numerical optimization and testing of fluid and configuration will be used.

Two configurations are accepted; the system in Figure 3(a) is a subcritical ORC while the system in Figure 3(b) is a supercritical ORC. In both configurations, a recuperator can be installed. This component allows the expander output to heat up the pump output. This component is used whenever $T_6 > T_2$. Furthermore, the optimal expander inlet pressure indicated during the simulation determinates whether the ORC will operate under supercritical condition or not.

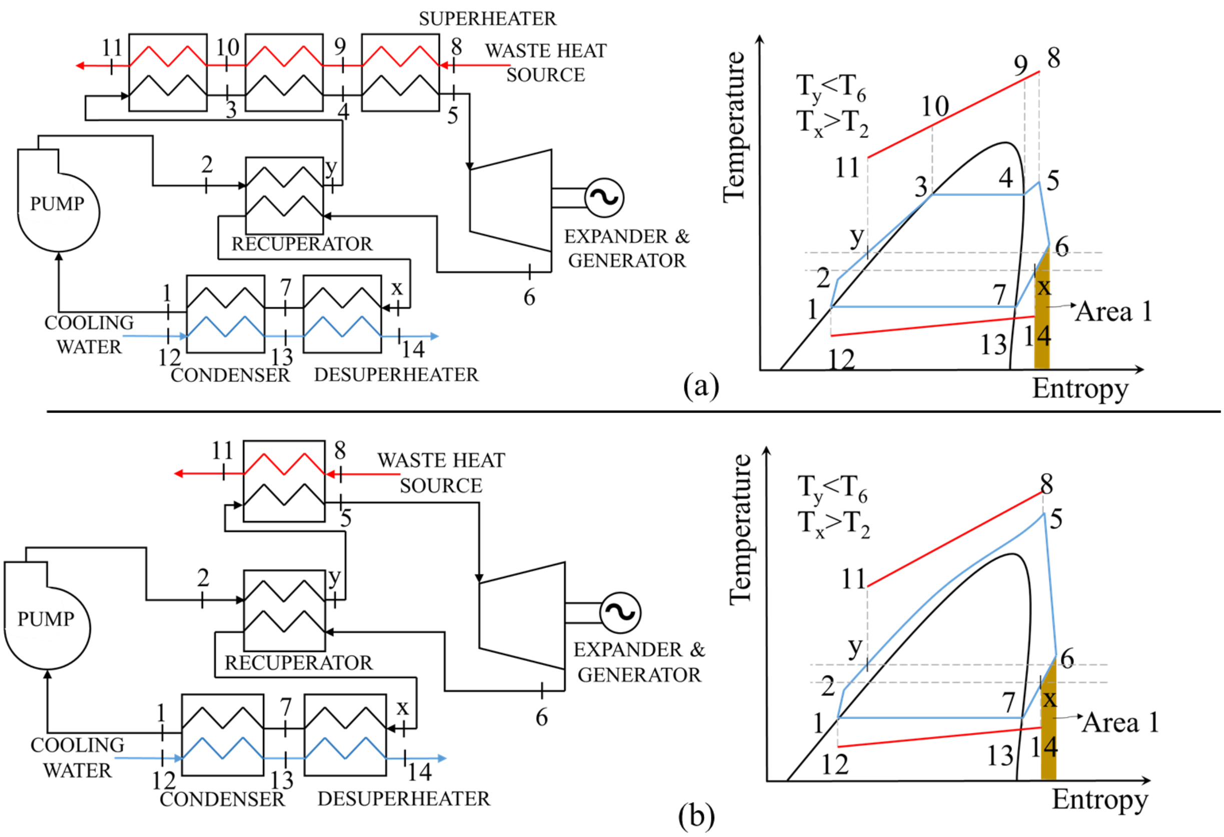

Figure. 3. Configuration and T-s diagram of ORC: a) subcritical, b) supercritical.

The components of the proposed system are modeled at steady state, potential and kinetic energy variations are neglected, and pressure drop along pipes and heat exchangers are disregarded. Table 2 indicates the main parameters used for modeling while Fig. 4 indicates the sequence of steps followed for fluid and configuration selection and operating condition optimization. Firstly, an organic fluid is chosen, and then T5 and P5 which are condition at turbine inlet are optimized defining the working fluid mass flow rate since the quantity of heat to be recovered is known. The net power output is the objective function to be maximized. Point 6 can be calculated since the condensation temperature and the isentropic efficiency of the turbine are defined. The thermal scheme with recuperator is tested if the temperature at point 6 is higher than at point 2.

Table 2. System Specifications.

Parameter	**Value**
Expander isentropic efficiency ($\eta_{t,ise}$)	80.0%
Pump Isentropic efficiency ($\eta_{p,ise}$)	75.0%
Recuperator effectiveness (ε_r)	85.0%
Pinch temperature difference at condenser	10.0°C
Minimum pinch temperature difference at evaporator	10.0°C
Exhaust gases temperature at state 8 (T_8) (stream 10 in engine plant)	231.18°C
Minimum exhaust gases temperature at state 11 (T_{11})	105°C
Exhaust gases mass flow rate ($\dot{m}_{eg}$) (stream 10 in engine plant)	21.42 kg/s
Exhaust gases average specific heat ($c_{P,eg}$)	1.20kJ/(kg·K)
Cooling water temperature at state 12 ($T_{12,dsw}$)	25.00 °C
Condensation temperature ($T_{7,dsw}$)	40.0 °C

The isentropic efficiency definitions for the expander and pump are given in Eqs. (6) and (7), respectively while Eqs. (8) and (9) show energy balance for expander and pump.

$$\eta_{t,ise} = \frac{h_5 - h_6}{h_5 - h_{6,ise}} \tag{6}$$

$$\eta_{p,ise} = \frac{h_1 - h_{2,ise}}{h_1 - h_2} \tag{7}$$

$$\dot{W}_t = \dot{m}_{wf} \cdot (h_5 - h_6) \tag{8}$$

$$\dot{W}_p = \dot{m}_{wf} \cdot (h_2 - h_1) \tag{9}$$

Equations (10), (11), (12) and (13) provide the energy balance for the economizer, evaporator, super-heater and supercritical evaporator.

$$\dot{m}_{eg} \cdot c_{P,eg} \cdot (T_{11} - T_{10}) = \dot{m}_{wf} \cdot (h_3 - h_2) \tag{10}$$

$$\dot{m}_{eg} \cdot c_{P,eg} \cdot (T_{10} - T_9) = \dot{m}_{wf} \cdot (h_4 - h_3) \tag{11}$$

$$\dot{m}_{eg} \cdot c_{P,eg} \cdot (T_9 - T_8) = \dot{m}_{wf} \cdot (h_5 - h_4) \tag{12}$$

$$\dot{m}_{eg} \cdot c_{P,eg} \cdot (T_{11} - T_8) = \dot{m}_{wf} \cdot (h_5 - h_2) \tag{13}$$

Energy balance for desuperheater and condenser are given in Eqs. (14) and (15).

$$\dot{m}_{cw} \cdot c_{P,cw} \cdot (T_{14} - T_{13}) = \dot{m}_{wf} \cdot (h_6 - h_7) \tag{14}$$

$$\dot{m}_{cw} \cdot c_{P,cw} \cdot (T_{13} - T_{12}) = \dot{m}_{wf} \cdot (h_1 - h_7) \tag{15}$$

Eq. (16) shows effectiveness definition for heat exchanger applied to the recuperator. Energy balance for this equipment is shown in Eq. (17).

$$\varepsilon_r = \frac{T_y - T_2}{T_6 - T_2} \tag{16}$$

$$h_y - h_2 = h_6 - h_x \tag{17}$$

Definition of degrees of superheat is given in Eq. (18). Thus, it is directly dependent on the optimized variable T5.

$$\Delta T_{sup} = T_5 - T_4 \tag{18}$$

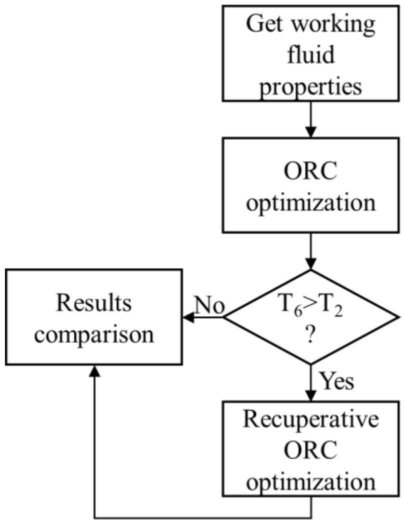

Figure 4. Simulation and optimization flowsheet.

Heat transferred from the exhaust gases $\dot{Q}_h$ is calculated by Eq. (19). Finally, net power $\dot{W}_{net}$ and cycle efficiency η_{orc} are given by Eqs. (20) and (21).

$$\dot{Q}_h = \dot{m}_{eg} \cdot c_{P,eg} \cdot (T_{11} - T_8) \tag{19}$$

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_p - \dot{W}_{p,cw} \tag{20}$$

$$\eta_{orc} = \frac{\dot{W}_{net}}{\dot{Q}_h} \tag{21}$$

Six organic working fluids and water were considered for the proposed system. These fluids were selected based on similar applications such as [42], [23] and [43]. The layouts described were simulated and optimized. The independent variables (T5 and P5) boundaries are set in order to avoid unrealistic values. Simulations were conducted on software EES [44]. Table 3 gives the list of the working fluids considered, their properties taken from the software library and the reference for the equation of state (EOS) used. The optimization is based on a built-in genetic algorithm (GA) available on EES. The genetic algorithm is based on Charles Darwin's theory of evolution and designed to reliably locate a global optimum even in the presence of local optima. Initially, a population of individuals (possible solutions) is randomly chosen and the adaptability – objective function value – of each one is determined; after that, a new generation is obtained from the current population, whose fittest individuals are prone to pass on their characteristics to descendants. In addition to the selection of the fittest, mechanisms of crossover and mutation also guarantee the characteristics variability of descendants [41].

Table 3. Working Fluids Properties.

Substance	**Type**	**Tcrit (°C)**	**Pcrit (kPa)**	**EOS Reference**
Cyclohexane	Isentropic	281	4081	[45]
Cyclopentane	Isentropic	239	4571	[46]
R123	Isentropic	184	3668	[47]
R134a	Isentropic	101	4059	[48]
Novec649	Dry	169	1869	[49]
Ethanol	Wet	242	6268	[50]
Water	Wet	374	22060	Steam_IAPWS correlations

6. RESULTS FOR THE OPTIMIZED ORC

Optimal values for the independent variables (T_5 and P_5) as well as working fluid mass flow rate ($\dot{m}_{wf}$), degrees of superheat (ΔT_{sup}) are given in Table 4. It is worth noting that the value of P5, except for water and cyclohexane in some extend, is high enough to provide a reasonable flow rate through the turbine.

Table 4. Properties of optimized ORC.

Substance	T_5 (°C)	P_5 (kPa)	$\dot{m}_{wf}$ (kg/s)	ΔT_{sup} (°C)
Cyclohexane	137.5	427	6.854	0.0394
Cyclopentane	143.4	1027	6.658	0.4638
R123	221.2	3825	14.37	-
R134a	221.2	9506	14.24	-
Novec649	221.2	2949	23.63	-
Ethanol	221.1	522.9	2.548	94.36
Water	220.8	221	1.05	97.37

Ethanol and water were the only fluids for which the optimized condition provided T6 < T2 making the use of the recuperator impossible. For the other fluids a dry expansion is obtained which helps turbine maintainability since there will be no water droplets at the latest stages of the turbine. The lowest mass flow rate for the water plant (regular Rankine cycle) indicates that it tends to be the most compact plant; on the other hand it will require a partial vacuum in the condenser which means more components such as vacuum pumps and/or ejectors. This plant is less efficient, less power is generated and the turbine will operate poorly due to a low pressure variation between its inlet and outlet.

For most of the fluids the upper limit set for temperature (221.2°C) was reached. As indicated in Fig. 5, R123 and novec649 were the fluids that provided highest net power: 715.2 kW and 673.3 kW, respectively. R123 is a well-known non-flammable HCFC refrigerant fluid usually considered for replacement of R11 while the Novec649 is also a non-flammable fluid usually considered for replacement of HFCs refrigerants. The R123, R134a and Novec649 optimal conditions are supercritical which means that there will be no evaporation process and only one heat exchanger will be required in the boiler, Fig.3b. From these results, it is possible to conclude that there are indeed many substances that will perform better than water in recuperating low temperature heat for electricity production. Furthermore, the addition of an ORC to produce electricity from exhaust gases will add up to 715.2 kW of power without consuming any extra fuel, which means an efficiency improvement of 7.2% in overall efficiency of the power plant from 43.1% to 46.2%.

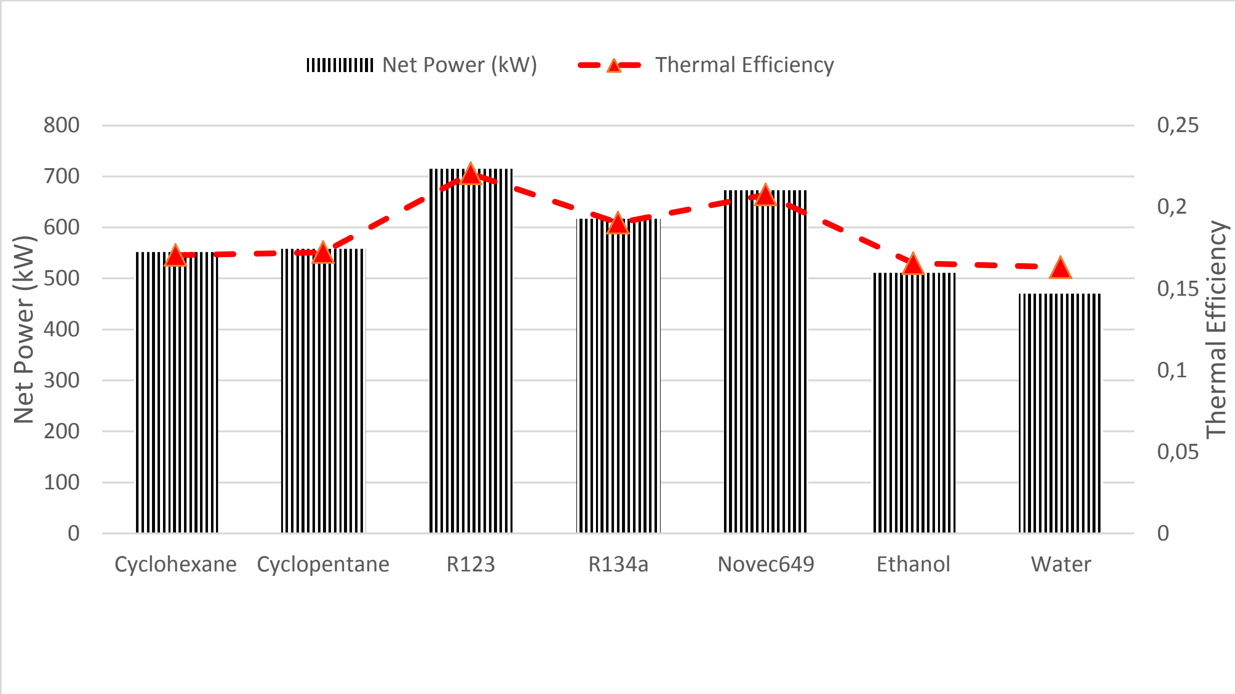

Figure 5. Optimized net power output and thermal efficiency results.

It worth noting that the main drawbacks regarding the use of ORCs coupled with reciprocating engines (lack of space, load variation and increase of back pressure) are less accentuated in stationary power plants in which the load is kept constant, there is plenty of space and blowers and stacks can be used to move the exhaust gases through the equipment.

7. CONCLUSION

A typical backup power plant using ci-engine-based technology was evaluated using exergy analysis. It was revealed that although almost 29% of energy input is sent to the environment by the cooling systems it represents only 2% of the exergy input. Furthermore, the engine internal exergy destruction represents about 44% of exergy input and it is unavoidable since it is inherent to the technology used. The exergy of the exhaust gases represent 6.5% of the exergy input and can be totally recovered for power production from thermodynamic point of view. Based on literature review, ORC was the technology chosen to make use of exhaust gases exergy. Recuperative and non-recuperative ORC configurations were tested for 7 working fluids selected from similar applications. The operating condition for these fluids was optimized, accepting super and subcritical combinations, and the results compared. The best fluids from thermodynamic point of view were the R123 and Novec649, both operating in supercritical condition and using a recuparator to preheat the working fluid. They provided extra 715.2 kW and 673.3 kW, respectively, with no additional fuel consumption which means an increase of 7.2% and 6.8%in the efficiency of the power plant, respectively. It represents an expressive increase that might be important especially when the capacity factor of these reciprocating internal combustion engines-based power plants increases.

ACKNOWLEDGMENTS

The first author would like to thank CAPES process number PGCI 88881.137765/2017-01 for the financial support.

NOMENCLATURE

b	specific exergy, kJ/kg
$\dot{B}$	exergy flow rate, kW
c	specific heat, kJ/(kg K)
EOS	equation of state
h	specific enthalpy, kJ/kg
LHV	low heating value, kJ/kg
$\dot{m}$	mass flow rate, kg/s
P	pressure, kPa
$\dot{Q}$	heat, kW
R	overall gas constant, kJ/(kmol.K)
s	specific entropy, kJ/(kg.K)
T	temperature, °C
x	mass fraction, %
y	molar fraction, %
$\dot{W}$	power, kW

Greek symbols

γ	activity coefficient
ε	heat exchanger effectiveness
η	efficiency
Δ	absolute variation
φ	chemical exergy to LHV ratio

Subscripts and superscripts

0	reference condition
ch	Chemical
cw	cooling water
eg	exhaust gases
h	hot source
ise	isentropic
mix	mixture
net	net
p	pump/constant pressure
ph	Physical
r	recuperator
st	standard
sup	superheat
t	turbine
wf	working fluid

REFERENCES

[1] International Energy Agency (IEA), "World Energy Outlook," 2017.

[2] E. de P. E. (Brasil), "Balanço Energético nacional 2016: Ano base 2015," *Empres. Pesqui. Energética - Rio janeiro EPE, 2016*, p. 292, 2016.

[3] Empresa de Pesquisa Energética, "Balanço Energético Nacional - Relatório Síntese," p. 61, 2017.

[4] Ibrahim Dincer; Marc A. Rosen, *Exergy*, 2nd ed. Elsevier Science, 2012.

[5] Szargut J., *Exergy Method, Technical and Ecological Applications*. WIT Press, 2005.

[6] T. J. Kotas, *The exergy method of thermal plant analysis*, vol. 20. Krieger Pub Co, 1985.

[7] S. de Oliveira, *Exergy: Production, cost and renewability*, vol. 63. Springer Science & Business Media, 2013.

[8] W. Stanek, J. Szargut, and L. Czarnowska, *Application of Thermo-ecological Cost (TEC) as sustainability measure for useful products*, no. 9783319486482. Springer, Cham, 2017.

[9] A. Tsolakis and S. E. Golunski, "Sensitivity of process efficiency to reaction routes in exhaust-gas reforming of diesel fuel," *Chem. Eng. J.*, vol. 117, no. 2, pp. 131–136, 2006.

[10] A. Tsolakis and A. Megaritis, "Catalytic exhaust gas fuel reforming for diesel engines - Effects of water addition on hydrogen production and fuel conversion efficiency," *Int. J. Hydrogen Energy*, vol. 29, no. 13, pp. 1409–1419, 2004.

[11] A. Tsolakis and A. Ã. Megaritis, "Exhaust gas assisted reforming of rapeseed methyl ester for reduced exhaust emissions of CI engines," *Biomass and Bioenergy*, vol. 27, pp. 493–505, 2004.

[12] A. Tsolakis, A. Megaritis, and D. Yap, "Application of exhaust gas fuel reforming in diesel and homogeneous charge compression ignition (HCCI) engines fuelled with biofuels," *Energy*, vol. 33, pp. 462–470, 2008.

[13] M. Herreros, A. Tsolakis, W. Wang, and A. P. E. York, "Ammonia as hydrogen carrier for transportation ; investigation of the ammonia exhaust gas fuel reforming," *Hydrog. Energy*, vol. 8, no. 0, 2013.

[14] A. Tsolakis and A. Megaritis, "Partially premixed charge compression ignition engine with on-board H 2 production by exhaust gas fuel reforming of diesel and biodiesel," *Int. J. Hydrogen Energy*, vol. 30, no. 7, pp. 731–745, 2005.

[15] U. Larsen, T. Van Nguyen, T. Knudsen, and F. Haglind, "System analysis and optimisation of a Kalina split-cycle for waste heat recovery on large marine diesel engines," *Energy*, vol. 64, pp. 484–494, 2014.

[16] M. Wang, X. Feng, and Y. Wang, *Comparison of Energy Performance of Organic Rankine and Kalina Cycles Considering Combined Heat Sources at Different Temperature*, vol. 40. Elsevier Masson SAS, 2017.

[17] P. Bombarda, C. M. Invernizzi, and C. Pietra, "Heat recovery from Diesel engines: A thermodynamic comparison between Kalina and ORC cycles," *Appl. Therm. Eng.*, vol. 30, no. 2–3, pp. 212–219, 2010.

[18] M. Zhao, M. Wei, P. Song, Z. Liu, and Z. Wang, "Effects of the ORC Operating Conditions on the Engine Performance for an Engine-ORC Combined System," *Energy Procedia*, vol. 105, pp. 662–667, 2017.

[19] S. Lion, C. N. Michos, I. Vlaskos, C. Rouaud, and R. Taccani, "A review of waste heat recovery and Organic Rankine Cycles (ORC) in on-off highway vehicle Heavy Duty Diesel Engine applications," *Renew. Sustain. Energy Rev.*, vol. 79, no. August 2016, pp. 691–708, 2017.

[20] V. Badescu, M. H. K. Aboaltabooq, H. Pop, V. Apostol, M. Prisecaru, and T. Prisecaru, "Design and operational procedures for ORC-based systems coupled with internal combustion engines driving electrical generators at full and partial load," *Energy Convers. Manag.*, vol. 139, pp. 206–221, 2017.

[21] G. Yu, G. Shu, H. Tian, Y. Huo, and W. Zhu, "Experimental investigations on a cascaded steam-/organic-Rankine-cycle (RC/ORC) system for waste heat recovery (WHR) from diesel engine," *Energy Convers. Manag.*, vol. 129, pp. 43–51, 2016.

[22] S. Lion, I. Vlaskos, C. Rouaud, and R. Taccani, "Overview of the Activities on Heavy Duty Diesel Engines Waste Heat Recovery with Organic Rankine Cycles (ORC) in the Frame of the ECCO-MATE EU FP7 Project," *Energy Procedia*, vol. 129, pp. 786–793, 2017.

[23] S. Seyedkavoosi, S. Javan, and K. Kota, "Exergy-based optimization of an organic Rankine cycle (ORC) for waste heat recovery from an internal combustion engine (ICE)," *Appl. Therm. Eng.*, vol. 126, pp. 447–457, 2017.

[24] R. Cipollone, D. Di Battista, and F. Bettoja, "Performances of an ORC power unit for Waste Heat Recovery on Heavy Duty Engine," *Energy Procedia*, vol. 129, pp. 770–777, 2017.

[25] C. Sellers, "Field operation of a 125kW ORC with ship engine jacket water," *Energy Procedia*, vol. 129, pp. 495–502,

2017.

[26] V. Apostol, H. Pop, A. Dobrovicescu, T. Prisecaru, A. Alexandru, and M. Prisecaru, “Thermodynamic analysis of ORC configurations used For WHR from a turbocharged diesel engine,” *Procedia Eng.*, vol. 100, no. January, pp. 549–558, 2015.

[27] J. Song, Y. Song, and C. Gu, “Thermodynamic analysis and performance optimization of an Organic Rankine Cycle (ORC) waste heat recovery system for marine diesel engines,” *Energy*, vol. 82, pp. 976–985, 2015.

[28] F. Yang, H. Cho, H. Zhang, and J. Zhang, “Thermoeconomic multi-objective optimization of a dual loop organic Rankine cycle (ORC) for CNG engine waste heat recovery,” *Appl. Energy*, vol. 205, no. 100, pp. 1100–1118, 2017.

[29] T. Chen, W. Zhuge, Y. Zhang, and L. Zhang, “A novel cascade organic Rankine cycle (ORC) system for waste heat recovery of truck diesel engines,” *Energy Convers. Manag.*, vol. 138, pp. 210–223, 2017.

[30] G. Shu, M. Zhao, H. Tian, H. Wei, X. Liang, Y. Huo, and W. Zhu, “Experimental investigation on thermal OS/ORC (Oil Storage/Organic Rankine Cycle) system for waste heat recovery from diesel engine,” *Energy*, vol. 107, pp. 693–706, 2016.

[31] G. Shu, G. Yu, H. Tian, H. Wei, X. Liang, and Z. Huang, “Multi-approach evaluations of a cascade-Organic Rankine Cycle (C-ORC) system driven by diesel engine waste heat: Part A – Thermodynamic evaluations,” *Energy Convers. Manag.*, vol. 108, pp. 579–595, 2016.

[32] M. Zhao, M. Wei, P. Song, Z. Liu, and G. Tian, “Performance evaluation of a diesel engine integrated with ORC system,” *Appl. Therm. Eng.*, vol. 115, pp. 221–228, 2017.

[33] M. Zhao, M. Wei, G. Tian, and P. Song, “Simulation of effects of ORC system installation on heavy-duty truck,” *Appl. Therm. Eng.*, vol. 128, pp. 1322–1330, 2018.

[34] V. Chintala, S. Kumar, and J. K. Pandey, “A technical review on waste heat recovery from compression ignition engines using organic Rankine cycle,” *Renew. Sustain. Energy Rev.*, vol. 81, no. June 2016, pp. 493–509, 2018.

[35] J. A. M. da Silva, S. Ávila Filho, and M. Carvalho, “Assessment of energy and exergy efficiencies in steam generators,” *J. Brazilian Soc. Mech. Sci. Eng.*, vol. 39, no. 8, pp. 3217–3226, 2017.

[36] J. Szargut, “Analysis of cumulative exergy consumption,” *Int. J. Energy Res.*, vol. 3, no. 4, pp. 541–547, 1987.

[37] T. J. Kotas, “Nomenclature for exergy analysis (Kotas 1995).pdf,” vol. 209, no. March 1994, 1995.

[38] ISO-3046-1:2002, “International Standard,” *Reciprocating internal combustion engines — Performance —Part 1*. pp. 1–30, 2002.

[39] G. Tsatsaronis, T. Morosuk, D. Koch, and M. Sorgenfrei, “Understanding the thermodynamic inefficiencies in combustion processes,” *Energy*, vol. 62, pp. 3–11, 2013.

[40] V. R. Seifert, Y. M. Barbosa, J. A. M. da Silva, and E. A. Torres, “Rankine cycle power augmentation: a comparative case study on the introduction of ORC or absorption chiller,” *J. Brazilian Soc. Mech. Sci. Eng.*, vol. 39, no. 11, pp. 4837–4848, 2017.

[41] B. M. Mazetto, J. A. M. Silva, and S. Oliveira, “Are ORCs a good option for waste heat recovery in a petroleum refinery?,” *Int. J. Thermodyn.*, vol. 18, no. 3, 2015.

[42] C. Gracco, J. Augusto, and S. De Oliveira, “Deep water cooled ORC for floating oil platform applications,” *Int. J. Thermodyn.*, vol. 20, no. 4, pp. 229–237, 2017.

[43] R. Scaccabarozzi, M. Tavano, C. M. Invernizzi, and E. Martelli, "Thermodynamic Optimization of heat recovery ORCs for heavy duty Internal Combustion Engine: Pure fluids vs. zeotropic mixtures," *Energy Procedia*, vol. 129, pp. 168–175, 2017.

[44] S. A. Klein, "EES - Engineering Equation Solver." 2017.

[45] Y. Zhou, J. Liu, S. G. Penoncello, and E. W. Lemmon, "An Equation of State for the Thermodynamic Properties of Cyclohexane," *J. Phys. Chem. Ref. Data*, vol. 43, no. 4, 2014.

[46] M. Thol, F. H. Dubberke, E. Baumhögger, R. Span, and J. Vrabec, "Speed of Sound Measurements and a Fundamental Equation of State for Hydrogen Chloride," *J. Chem. Eng. Data*, vol. 63, no. 7, pp. 2533–2547, 2018.

[47] R. Tillner-Roth, "Fundamental equations of state." Aachen: Shaker-Verlag, 1998.

[48] R. Tillner Roth and H. D. Baehr, "An International Standard Formulation for the Thermodynamic Properties of 1,1,1,2-Tetrafluoroethane (HFC-134a) for Temperatures from 170 K to 455 K and Pressures up to 70 MPa," *J. Phys. Chem. Ref. Data*, vol. 23, no. 5, pp. 657–729, 1994.

[49] M. O. McLinden, R. A. Perkins, E. W. Lemmon, and T. J. Fortin, "Thermodynamic properties of 1,1,1,2,2,4,5,5,5-nonafluoro-4-(trifluoromethyl)-3-pentanone: Vapor pressure, (p, ρ, T) behavior, and speed of sound measurements, and an equation of state," *J. Chem. Eng. Data*, vol. 60, no. 12, pp. 3646–3659, 2015.

[50] J. A. Schroeder, S. G. Penoncello, and J. S. Schroeder, "A fundamental equation of state for ethanol," *J. Phys. Chem. Ref. Data*, vol. 43, no. 4, 2014.